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DETAILS OF THE CONSTRUCTION AND PRODUCTION OF FUEL PUMPS AND FUEL NOZZLES FOR THE AIRPLANE DIESEL ENGINE

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DETAILS OF THE CONSTRUCTION AND PRODUCTION OF FUEL PUMPS AND FUEL NOZZLES FOR THE AIRPLANE DIESEL ENGINE*

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Compressorless diesel engines of the stationary, marine, locomotive, and automobile types have been manufactured in our plants for the past ten years. These engines are provided with pumps and nozzles produced in the same plants. Light diesels for land vehicles and airplanes are generally provided with imported fuel pumps and nozzles chiefly from the Bosch firm. For some years the question of the practicability of the home manufacture of fuel pumps and nozzles for this type of diesel was considered. This question began to be solved in a practical way after the first soviet diesel conference. Some years previously ZIAM had solved the problem by constructing a series of fuel pumps and nozzles.

The author, who is working with ZIAM in the construction and testing of fuel pumps and nozzles, considers it would be of some interest to impart within the limits of this short article some information on this work. Without entering into a consideration of the foreign construction of fuel pumps of the slide valve type, we shall touch only upon some of the work that is being done in our country.

Due to the simplicity of their construction (absence of intake and outlet valves), their compactness and small weight, the slide-valve pumps found wide application to high speed diesel engines. The closed type of nozzle was widely applied for high speed diesels because of the accuracy of their injection characteristics. The accuracy of the fuel injection, that is, the closeness with which the actual fuel discharge diagram follows the theoretical one, is a factor of prime importance. By the theoretical fuel injection diagram is meant the diagram constructed on the basis of the physical properties of the fuel from the time of opening to the closing of the valve. The work on the ZIAM pumps and nozzles was undertaken for the purpose of establishing an injection system for aviation diesel engines.

*Dieselestroyenie no. 6, Moskva, 1935, pp. 8-14.

Figure 1 shows a pump constructed by ZIAM under the direct supervision of the personnel of the institute. The names of the parts are given below the figure. During the construction of the pump it was known that the edge of the MAN pump was deformed and there were many repairs to be made.

In order to simplify the technical process and obtain a clear cut-off of the motion of the plunger, the valve of the ZIAM pump was first constructed in the form of a double spiral in the cylinder of the pump. On figures 2a and 2b are shown the openings of the valve with spiral edges, regulating the discharge from the pump which depends on the engine load. In constructing the valve the form and dimensions were so chosen that three borings could be made which could be united to form a 3-mm (0.118 in.) port in the valve. The valve of the ZIAM pump was likewise so constructed that the grinding of the cylinder could be made more easily than the grinding of the pump plunger, and the sharp edges of the valve, pressed against the inside surface of the cylinder, would remain untouched.

Since the cylinder body is weakened by the cutting of the ports, it was possible, notwithstanding the symmetry of the arrangement, for the cylinder to be warped on calking. For these reasons it was decided to choose tool steel of type EO for the plunger and cylinder, as the most resistant against deformation during heat treatment. In this pump the outlet ports were separate from the inlet ports.

With the object of decreasing the amount of clearance of the pump and testing the effect of the friction on the moving parts of the governor on the cut-off force, the plunger spring was held against a toothed crown and collar of the plunger. Afterwards the lower part of the plunger and spring seat were changed as shown in figure 1.

The pump underwent 650 hours of continuous testing. The tests were carried out on a test stand and on the engine after the characteristic curves of discharge were obtained.

In the process of production, of assembly, and of tightness and pump tests, the following points were brought out:

<u>Production</u>.- 1) The warping of the cylinder on calking

was practically negligible.

2) The inner edges of the valve were not deformed in the cutting process and during grinding; as a result, the first set of cylinders and plungers was without defect.

3) The deflection of the spiral coordinate amounted to 0.2 to 0.5 mm (0.0079 in. to 0.0197 in.), that is, the discharge openings did not open at the same time, but with one plunger from 0.5 to 1.5 cam shaft degrees behind the other, depending on the cam profile.

4) The toothed governor engaged well with the toothed crown.

5) No difficulty was found in the construction of the remaining parts of the pump (the discharge valve, springs, and seat of the discharge valve) as originally designed.

Assembly.- During assembly the following defects in the construction were revealed:

1) The groove in the toothed crown for guiding the cross head of the plunger (fig. 1, detail 7) was cut slantwise, that is, the cylinder forming the toothed crown was not strictly parallel to the plunger axis; as a result, the plunger cut into its guide. This defect was corrected by using another toothed crown.

2) It was necessary to reduce the seat of the pump cylinder (detail 1) by 15 microns.

3) During strong compression of the stem of the discharge valve the plunger cylinder underwent a very small deformation because of the many cut-outs in it, and the return motion of the plunger became difficult.

Leakage tests.- The leakage tests were carried out with solar oil and gas oil. Figure 3 shows the results of the leakage test on the pump using solar oil and the results are compared with those obtained from tests on heavy diesels. On the same figure are also drawn curves for the Bosch and Benes pumps. The leakage tests give an indication of the amount of plunger clearance, the tightness of the pump discharge chamber, the departure of the several parts from the circular and cylindrical forms, and the deformation of the valve edges. The hydraulic test method is the most suitable for bringing out these factors, since it makes it possible to determine exactly any defects in the construction of the pump plunger and cylinder.

There is no better method for determining exactly the inside diameter of the pump cylinder than that employing fluid-measuring apparatus. The tightness of the fit is characterized, not by the force with which the plunger is moved in the cylinder, but by the amount of sealing of the pressure chamber. It may happen that a tightly fitting plunger and cylinder are less well sealed than a freely moving plunger that has a strictly cylindrical form and is well ground.

The criterion for the tightness of the discharge chamber is the rate of fall of pressure of the compressed fluid:

$$V_p = \tan \alpha = \frac{dp}{dt} \tag{1}$$

where α is the inclination of the pressure curve to the time axis. Only in this way is it possible to compare the quality of manufacture of corresponding parts of a pump. From the rate of fall of pressure and the amount of fuel which leaks through it is possible to determine the effective cross sectional area of the clearance between the cylinder and the plunger, and from the area of the clearance to determine the inside diameter of the pump cylinder in the region of the valve. The clearance between the plunger and the cylinder is

$$e = \frac{d_2 - d_1}{2} \mu$$

where d_2, diameter of the cylinder,

d_1, diameter of the plunger,

μ denotes microns.

The clearance between the plunger and the cylinder defines the fitting of these parts. If the fuel were an ideal fluid and if the pump were to maintain a pressure within the range

p_2 to p_1 atmospheres, then for large diameters of the plunger the following expression would be obtained for the flow through the clearance:

$$\frac{d_1}{d_2} = \sqrt{1 - \frac{2 \times 10^2 (\sqrt{p_1} - \sqrt{p_2})}{\mu\ t \sqrt{2\ g} \sqrt{\gamma}}} \qquad (2)$$

where μ is the discharge coefficient taken as unity

γ, specific weight of the fuel

g, 9.81 meters/sec.^{2}

p, pressure in atmospheres

The flow of an ideal fluid under a difference in pressure heads is given by

$$t = \frac{2\ \Omega\ [\sqrt{H_1} - \sqrt{H_2}]}{\mu\ \omega\ \sqrt{2g}}$$

where t is time of discharge,

Ω, area of vessel,

H_1 and H_2, pressure at beginning and end of discharge,

ω, cross sectional area of opening,

μ, coefficient of discharge.

Formula (2) does not take the fuel viscosity into account or the capillary properties of the clearance and it therefore cannot be used for determining the effective clearance, the Poiseuille formula being required. If the volume of the fluid under pressure is V_0 cm^3 then for a pressure fall from p_1 to p_2 atm. the volume of fuel leaking through is equal to:

$$\alpha\ \Delta\ p\ V_0$$

The discharge per second through the clearance is

$$\frac{\alpha \, \Delta p \, V_0}{\Delta t} = F \, W = \frac{p_1 + p_2}{2} \times \frac{1}{128 \, \mu \, l} \pi \, d^4$$

where μ is absolute viscosity in kg sec./cm^2,

l, length of surface in cm,

F, clearance in cm^2,

d, diameter of clearance,

$\frac{p_1 + p_2}{2}$, fuel pressure drop in leaking through clearance

α, coefficient of compressibility of fuel $\frac{1}{20000}$ cm^3/atm.

Since $\frac{dp}{dt} = V_p$ is the rate of pressure drop in the tightness test and p denotes the pressure in the system,

$$\alpha \, V_o \, V_p = \frac{p}{128 \, \mu \, l} \times \pi \, d^4$$

The inside diameter of the cylinder is accordingly found from the expression:

$$d = \sqrt[4]{\frac{\alpha \, V_o \, V_p \, 128 \, \mu \, l}{\pi \, p}} \qquad (3)$$

where $V_p = \frac{500 - 400}{180} = 0.555$ atm./sec.

$V_o = 100$ cm^3 (including volume of pressure)

$l = 0.4$ cm

diameter of plunger = 12 mm

diameter of cylinder = 12.003 mm

In testing the pump for leakage it is first necessary to check the tightness of the press and introduce the corresponding correction into the result. Thus the tightness of the discharge chamber of the pump depends on the clearance between the plunger and the cylinder, that is, on the nature of the rubbing surfaces and the nominal dimensions. Some engineers claimed that it was possible to fix accurately the dimensions of the pump cylinder, the plunger, and the nozzle needle by careful polishing.

Figures 10 to 14, section 8, part 1, of Prof. A. P. Sokolovsky's book, "Construction of Machinery" gives a graphic presentation of the dependence of the condition of a surface on the manner in which it is treated and also the effect of the condition of the surface on the life of the working parts. After polishing there remain sharp "little combs" on the pump cylinder as well as on the plunger, as a result of which the clearance between the plunger and the cylinder increases by the height of these combs, which appear as traces marked out by the polishing stone (fig. 4, a).

On fitting the plunger into the cylinder (especially when the fitting is tight) the projection A on the plunger removes the projection B on the cylinder; as a result there is a wearing away of the material which in the form of a barely noticeable stream fills up the clearance and spreads throughout, the surfaces in contact under the effect of the high pressure. By grinding (fig. 4, b) these irregularities of the surface are removed, the surfaces in contact are smoothed out, the effective clearance between the plunger and the cylinder is decreased, and the tightness of the fitting and sealing of the discharge chamber increased.

The importance of grinding for the cylinder, plunger, and needle of the closed type nozzle is confirmed by the fact that, when the polishing has not been done carefully and accurately enough, when deep traces of the polishing stone remain on the surface of the plunger or on the needle of the valve, the plunger or the nozzle needle cut in.

Figure 5 shows the surface of a needle of a closed valve at the place where it is cut in. The appearance of the surface where it is eaten in is sharply different from its appearance after grinding. The direction of the regions of light on the photograph where the surface is eaten in is parallel to the axis of the needle.

Figure 6 shows a photograph of a carefully ground surface. The figure shows no deep traces of the polishing stone; these traces are not deep and have the direction of a spiral of small pitch, the traces from the grinding are also directed along spiral lines, but of larger pitch. The ground surface is smooth without projecting irregularities.

The grinding is best done in the following order: For the hard tool steels (EU - 12) used for the pump plunger or the nozzle needle, the initial coarse grinding is done with red copper and the final grinding is done with the aid of cast iron grinders cut in the form of spirals.

The leakage tests of high speed diesels is made with the same fuel that is used by the engine at a pressure of 250 to 300 atmospheres.

<u>Testing the pump and nozzles</u>.- The pump is given a long run test to determine its reliability, and to obtain the characteristics of the injection, the coefficient of discharge, and the wear, and to reveal any defects in the construction.

The pump described in this article and shown on figure 1 underwent 500 hours of continuous running and 150 hours of testing on the engine. Measurements were taken every 150 hours.

<u>1. Cam profile</u>.- A series of cams was tried with the pump. The shape of cam no. 1 was tangential, that is, the eccentricity of the cam was equal to infinity. For the other profiles the curvature of the cam was gradually increased, thus decreasing the eccentricity. A prolonged test was carried out on a cam having a profile form analogous to that of the Bosch pump.

<u>2. Coefficient of discharge of the pump</u>.- Figure 7 shows the change in the discharge coefficient before the 500-hour test (curve 1) and after the 500-hour run (curve 2).

The wear of the plunger is shown by curve 3. The wear on the plunger is considerable because of the tight fitting. It appeared later on that for a high-speed-engine tight fitting is not necessary nor is a high pressure necessary, since curves 1 and 2 for the discharge coefficient tend to approach each other at the high speeds, and the higher the speed the less the effect of the clearance, which of course must not exceed a certain limit beyond which the loss in fuel for the engine begins to increase.

The drop in the discharge coefficient with increase in speed of cam shaft is explained not only by the increased injection pressure varying almost as the square, but also by the factor of the forced cut-off which depends on the dimensions of the discharge port and on the speed of the plunger motion from the beginning of fuel cut-off to the end of the fuel discharge. The discharge coefficient is affected by the following elements in the construction and manufacture of the pump:

1) The diameter of the plunger of the pump,

2) The amount of grinding,

3) The construction of the discharge valve,

4) The separation between the inlet and outlet valves,

5) The cam profile,

6) The construction and characteristics of the nozzle, (the stiffness of the nozzle spring, open or closed type of nozzle,

7) Length and diameter of the intake and discharge pipes of the pump,

8) Construction, size, position, and condition of the fuel filter,

9) The volume of the system and the fuel pressure.

Characteristics of the injection.- Figure 8 shows the injection characteristics before and after the 500-hour test. After prolonged tests the working of the pump undergoes the following changes:

1) The beginning of injection is somewhat delayed,

2) The time discharge curve assumes a wave character,

3) The end of the discharge occurs somewhat earlier,

4) The leakage factor becomes of greater importance.

During the tests a series of pump valves was checked. In making changes in the construction of the valve, the following considerations were taken into account:

1) The desirability of decreasing the effect of microdeformations of the cylinder, weakened by the double cut-outs,

2) The necessity for simplifying the technical process of manufacture of the cylinder and plunger,

3) The desire to determine the effect of the valve construction on the character of the injection.

Figure 9 shows the changes in valve no. 1, introduced by the author of this article. On the cylinder 1 is placed a sleeve 2, on the projection of which, in the direction of the arrow, the cylinder is supported in the body of the pump. In this way the cylinder valve 3 is freed from axial loads. Moreover, between the cylinder and the sleeve there is formed a new chamber which functions as a hydraulic damper, and the fuel is led through the opening 5. The sleeve is pressed down on its seat.

The pump cylinder in its first form required a large amount of machining and a corresponding supply of high-speed machine tools. By a special production method (for example, that of the Brice firm) the cutting of the two ports of the slide valve could be accomplised in about 5 to 10 minutes. The Brice pump, which appeared about one year after the production of the ZIAM pump, had a similar construction of the valve in the pump cylinder but could be produced much faster. The changes brought about in the construction of the slide valve are shown on figure 10.

As shown on the figure the plunger underwent the following changes: instead of a single round opening for the inlet, two spiral grooves of 1.2-mm (0.047 in.) depth and 3-mm (0.118 in.) width were cut out on the plunger. During the testing of the first variant of the slide valve (fig. 2a) and the variant of figure 10 the inlet port was made round and in the form of a parallelogram with angle of 45° between the sides, described about a circumference 3 mm (0.118 in.) in diameter. For a double valve with a spiral running in one direction, a change in the inlet opening from the parallelogram form is possible only for a pitch angle of the valve spiral of 45°.

Figure 11 shows schematically the results of the tests of the valves of the different constructions and also the effect of the form of the inlet port on the end of the injection. Under certain conditions a pump with positive cut-off may work very satisfactorily without a discharge valve. The principle has this advantage, that it does not permit repeated openings of the needle of the closed nozzle, since, after cut-off in the valveless system there are no pressure waves in the fuel system. The pressure fluctuations in the discharge line after cut-off arise from the discharge valve on its seat and therefore if we eliminate the discharge valve we do away with the source of these fluctuations. In figure 12 curves 2 and 3 correspond to the fuel discharge without discharge valve.

The results of the pump tests on the engine showed that, with a good cut-off, accurate injection, assured by the proper adjustment of the pump elements, there is a decrease in the consumption of fuel and hence an increase in the rated power of the engine.

Translation by S. Reiss,
National Advisory Committee
for Aeronautics.

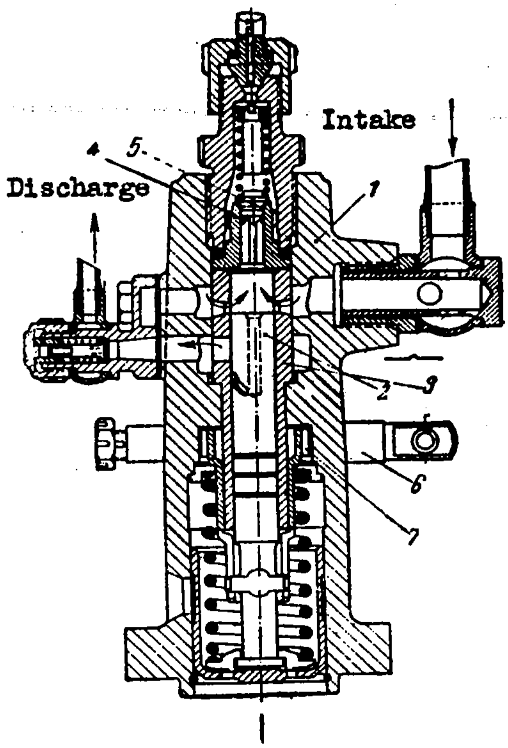


Figure 1.- Aviation diesel fuel pump. 1- body of pump constructed of electron. 2-plunger of EO steel. 3-cylinder of EO steel. 4- seat of discharge valve, of EU-12 steel. 5- discharge valve of EU-12 steel. 6 - toothed governor regulating amount of fuel discharge of U-4 steel. 7 - toothed crown of U-4 steel and dural.

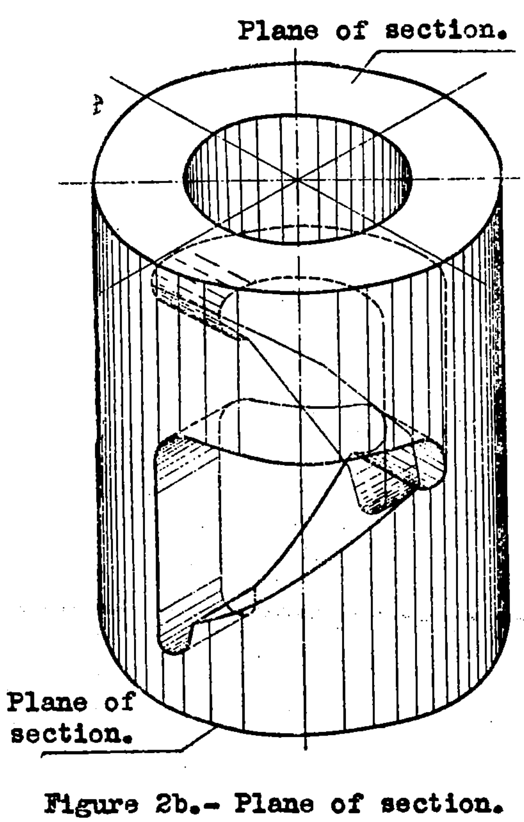


Figure 2b.- Plane of section.

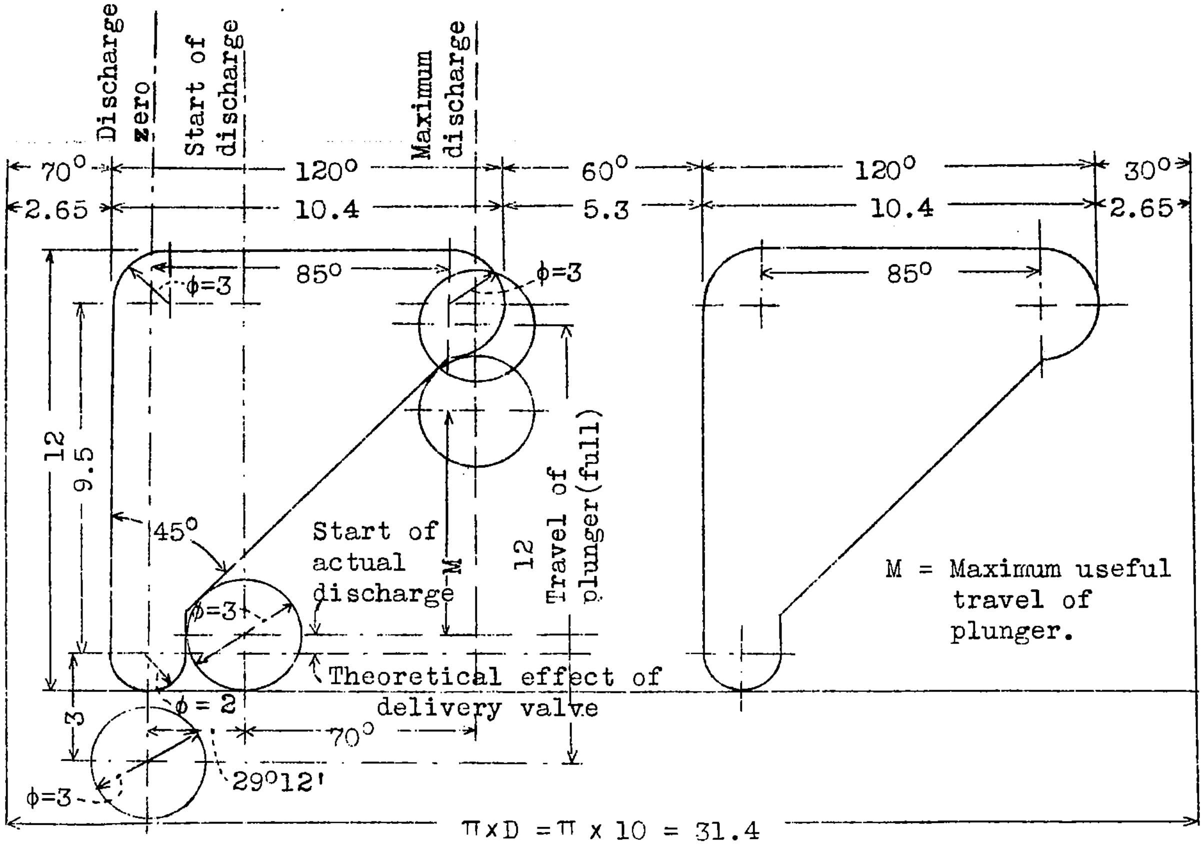


Figure 2a

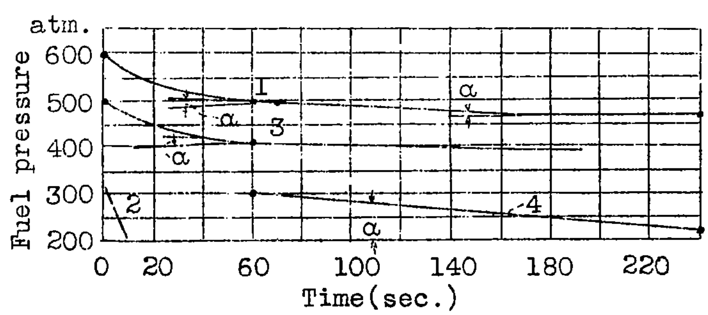


Figure 3

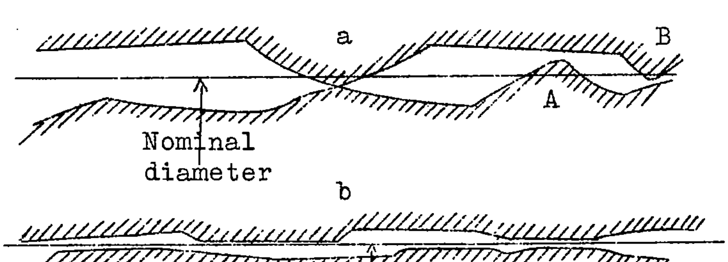


Figure 4

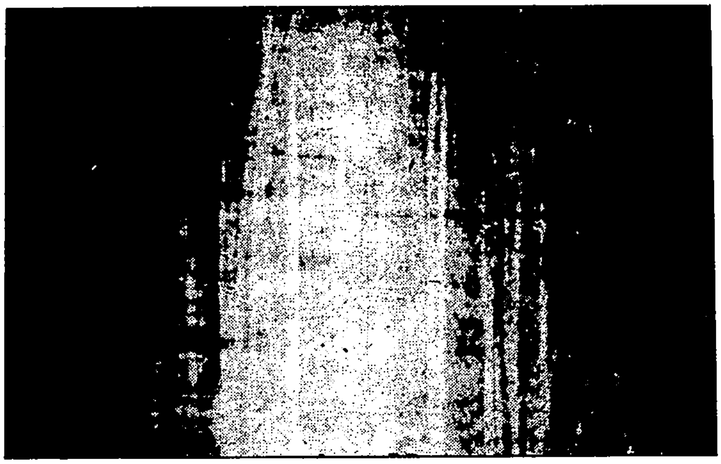


Figure 5.

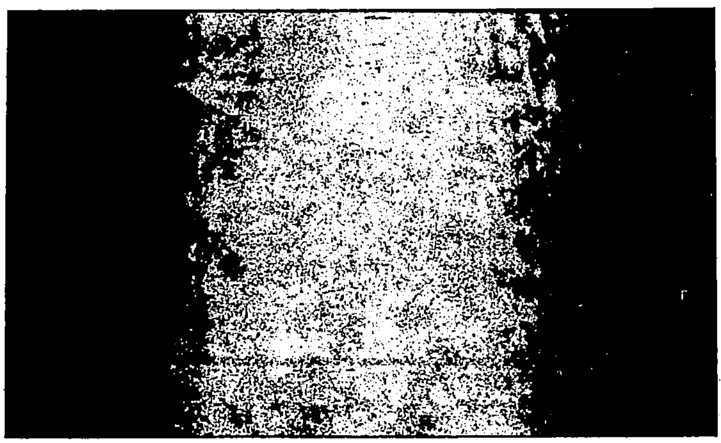


Figure 6.

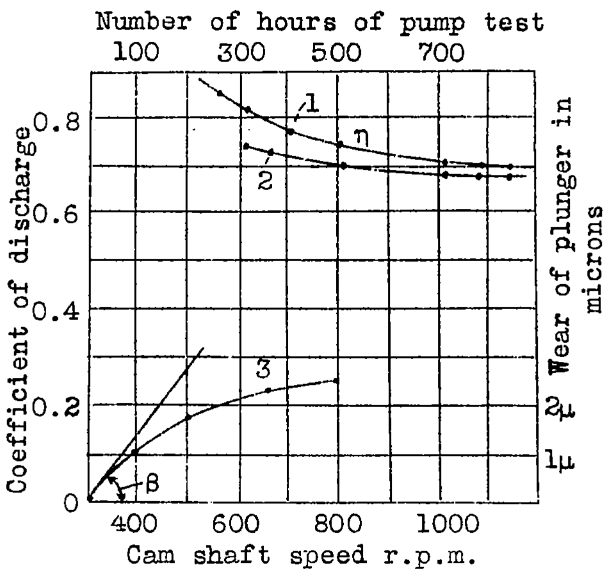


Figure 7

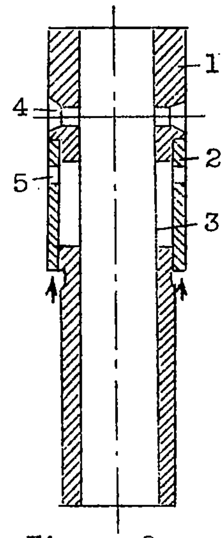


Figure 9

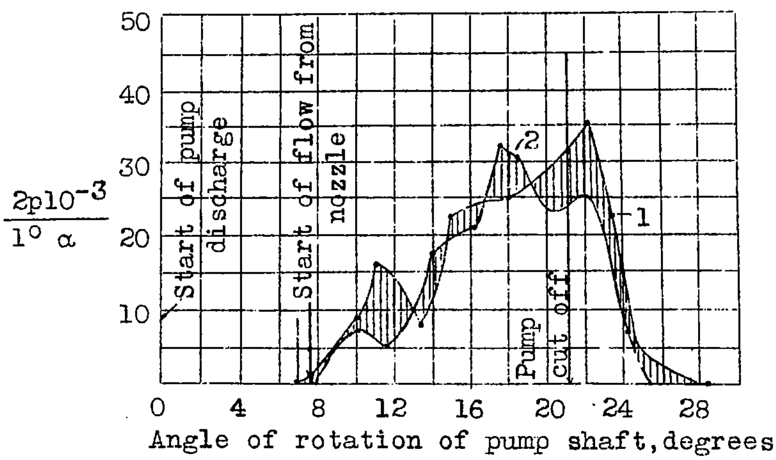
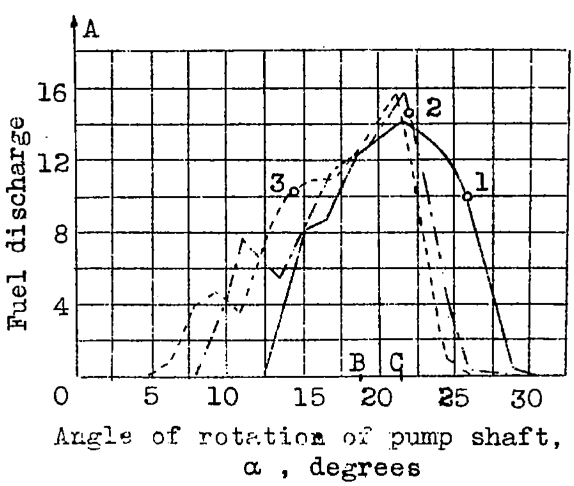


Figure 8



A. Theoretical start of discharge
B. Cut off for TN-1 pump
C. Cut off for Bosch pump

Figure 12

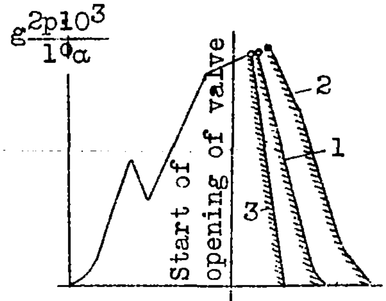
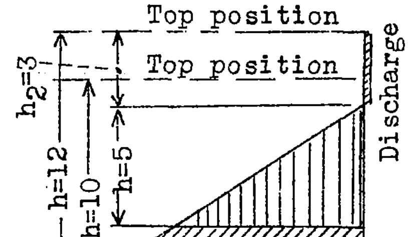
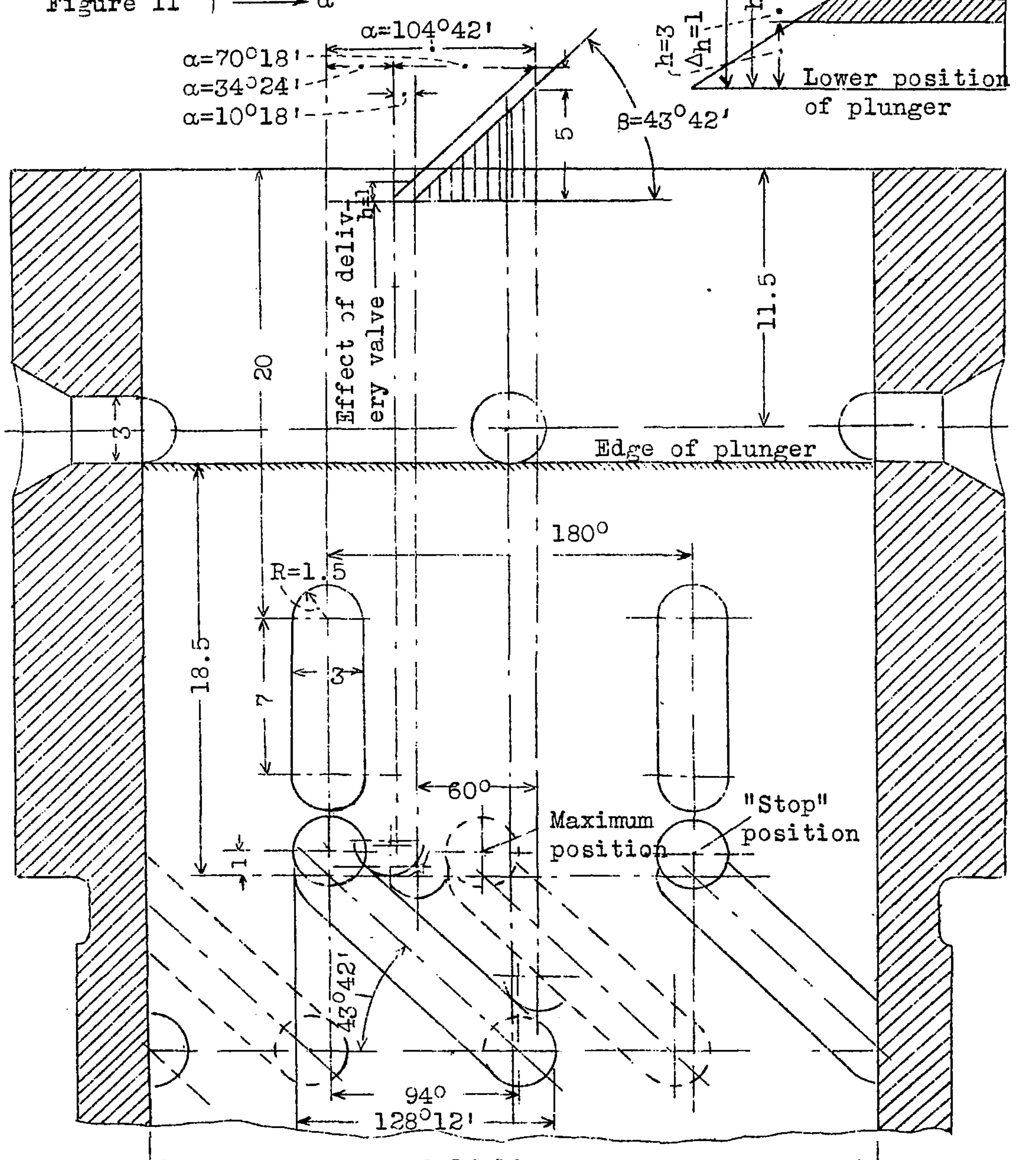


Figure 11

Pitch of spiral 30 mm. Left handed spiral
Pump discharge at 1800 r.p.m. Q = 16 kg /hr. q_T=340 mm ^{3}(per injection)
Figure 10.- Fuel pump valve developed.